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STRUCTURAL MODELING FOR CONTROL DESIGN
(Articulated Multibody Component Representation)

by
E. D. Haugse, R. E. Jones, and W. L. Salus
Boeing Aerospace and Electronics
Seattle, Washington

Abstract

High gain, high frequency flexible responses in gimbaled multibody systems are discussed. Their origin and physical significance are described in terms of detailed mass and stiffness modeling at actuator/sensor interfaces. Guyan Reduction, Generalized Dynamic Reduction, inadequate mass modeling detail, as well as system mode truncation, are shown to suppress the high gain high frequency response and thereby lose system flexibility important for stability and performance predictions. Model validation by modal survey testing is shown to risk similar loss of accuracy. Difficulties caused by high frequency responses in component mode simulations, such as DISCOS, and also linearized system mode simulations, are described, and approaches for handling these difficulties are discussed.

Introduction

The control-structure-interaction problem is concerned with locally applied input forces or torques and localized outputs at actuator/sensor interfaces. These localized inputs and outputs are usually modeled as occurring at single points or sections of structural members. This creates difficulties in regard to dynamic modeling, and careful flexibility and mass modeling at local input/output locations is required to accurately predict dynamic response. Local flexibility is often important because of mechanical details associated with actuators, sensors, and their mounting hardware. The frequencies of vibration at which the dynamic responses occur, characterizing the local flexibilities, depend on both the local flexibility and the mass modeling at the actuator/sensor interfaces. The mass and inertia at the interfaces are difficult to quantify, particularly for rotational degrees of freedom, and are often not done explicitly.

Typically the structural engineer will deliver Craig-Bampton component mode models, fixed at the actuator/sensor interfaces of a servo-mechanism, to the controls engineer. The controls engineer will then merge the component models, freeing the degrees of freedom associated with the control force or torque, and perform a system level simulation. This often results in transfer functions with high gain responses occurring at unexpected high frequencies. These high gain high frequency (HGHF) responses are due to vibration modes associated with small local interface mass and inertia connected to relatively stiff primary structure by flexible elements representing servo mechanisms and fastener schemes. If HGHF responses do not occur the modeling may be suspect for control-structure-interaction simulations.

This paper will address important aspects of flexibility and mass/inertia modeling of structure for the controls problem as they influence solvability of the equations of motion, accuracy of control-structure-interaction analyses, and testing procedures.

Local Flexibility and Mass Modeling

Multibody structures are connected by servo mechanisms that control the relative angle or displacement between one flexible body and another. The servo mounting hardware and its internal parts may cause significant loss of stiffness across the controlled joint. This is illustrated by the hardware schematic in Figure 1 in which a motor is shown attached to a bracket which is in turn attached to body A. The attachment scheme may use fasteners which add to the flexibility of the bracket.

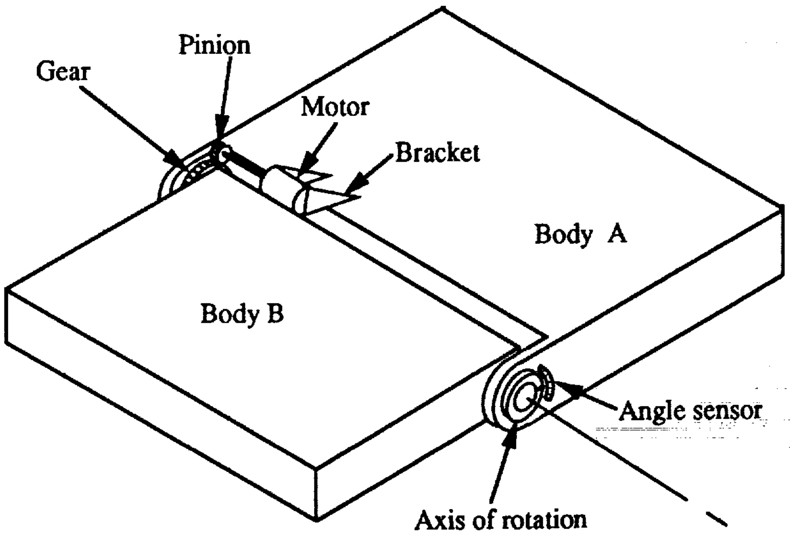
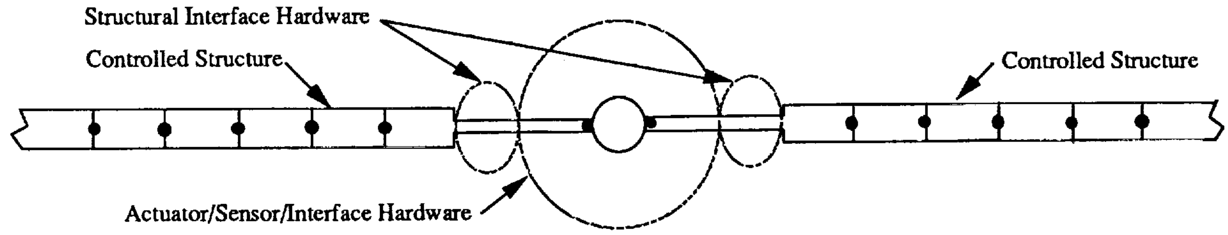


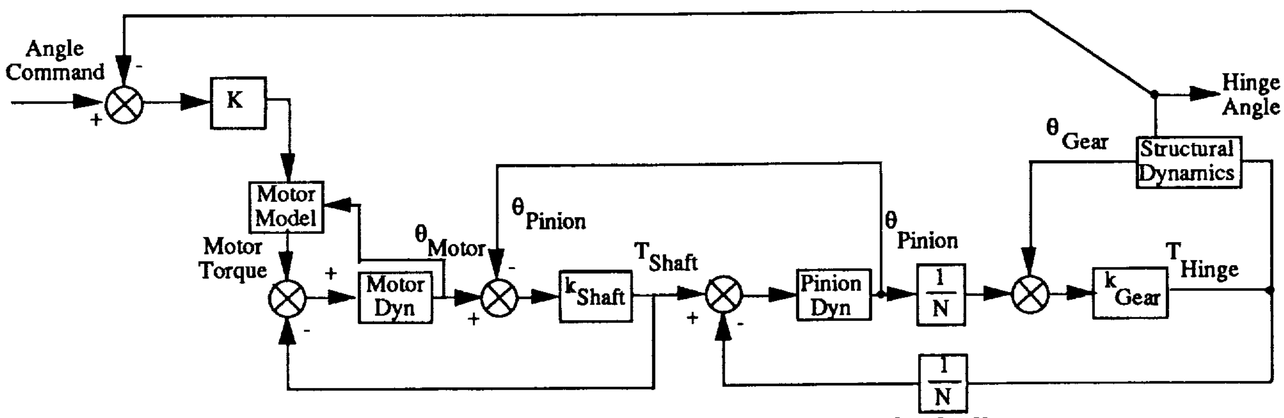
Figure 1. Hardware Schematic

Body B is connected to A through the gear contact forces, the pinion shaft, and some type of spline detail not shown on the figure. These are also sources of flexibility. This type of hardware can only be modeled accurately by a cooperative effort of structures and controls engineers. Some of the flexibility and mass will be modeled by the controls engineer, and some by the structures engineer. Therefore, modeling responsibilities need to be well defined to avoid exclusion or duplication of flexibility and mass/inertia. The structural engineer should understand the system block diagrams that the controls engineer will use in analysis. Similarly, the controls engineer needs to understand the types of structural modeling approximations which would reduce the accuracy of the analysis. To illustrate these points a simplified block diagram is shown in Figure 2.

In this example the controls engineer is responsible for modeling the motor shaft, gears, and the motor itself. The structures engineer is responsible for everything else including the motor bracket and fasteners. Body A should be supported at the motor rotor for modal analysis, and body B should be supported at the gear, thereby including gear spline flexibility effects. The block diagram shows that the driving torque at the hinge is determined from the difference between the rotations in two instances: the motor and the pinion; and the pinion and the gear. Each relative rotation is assigned a flexibility. This suggests that accuracy in computing local contributions to rotation is important. The block diagram illustrates the complexity of actuator-to-structure modeling and the degree to which the structures and controls engineers need to cooperatively build the dynamics simulation.



(a) Hardware schematic



(b) Simplified control - structure - interaction block diagram

Figure 2. Control - Structure Modeling

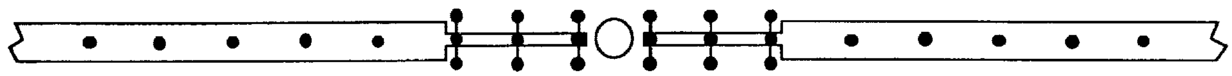
Local flexibility modeling may not suffice to provide the accuracy needed at actuator/sensor interfaces. Mass and inertia modeling are required to enable this flexibility in modal dynamics models. Since many actuators apply torque and sensors resolve angular motions, it is necessary to treat both rotational flexibilities and inertias very carefully, an area that many structural dynamicists do not pursue in detail when developing models. For example, adding any local inertia will enable the local flexibility, but may not properly define its frequency spectrum.

The vibration modes representing a large portion of the local flexibility often occur at very high frequencies. The high frequencies result from a small inertia supported by local flexibility. Figure 3 attempts to illustrate this.

In Figure 3a the model uses only lumped masses, while the model in Figure 3b includes inertias as well (at least at the hinged interface). The finite elements are intended to be small, aiming at an accurate modeling of local flexibility. Therefore, the masses are very small, and in Figure 3b the inertias are very small also. In the first case the modal analysis requires elimination of the rotational freedom at the hinge. This is done by Guyan reduction, and as a result the effect of the local flexibility is lost. In the second case the local inertias enable the local flexibility, all of which is present in the modal analysis. The small inertias, undergoing rotational modal motions supported by the attached elastic elements, will have very high frequencies. The actual values of these modal frequencies will depend on both the inertia values and the flexibility values of the model at the actuator interface. It may be worth noting that if the lumped mass model is solved for vibration modes without Guyan reduction, numerical error (round-off) may enable the local flexibility, producing extremely high eigenvalues. The accuracy of this numerical process is questionable, especially for controls applications, because it may affect the placement of low frequency transfer function zeros.

- Total flexibility not obtained

(a) Lumped mass model (Guyan reduction at interface)

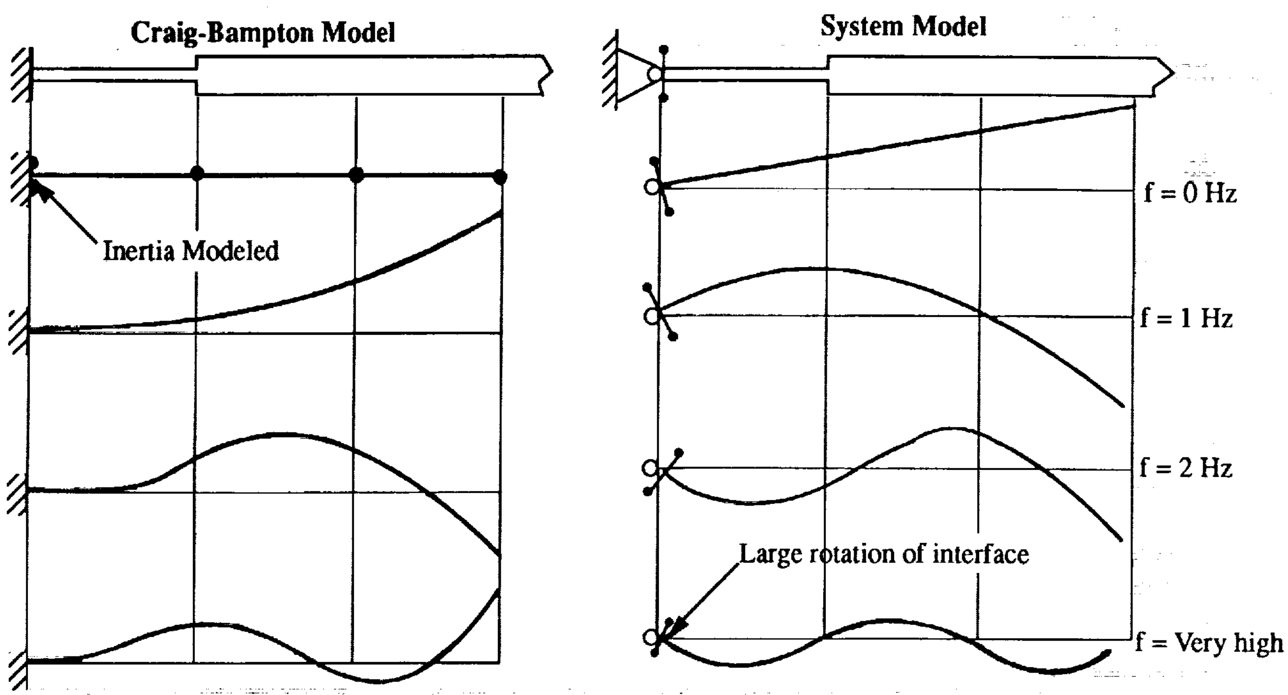


- Total flexibility obtained

(b) Model with inertias

Figure 3. Comparison of Models With and Without Inertia

It is reasonable to question how, starting with a Craig-Bampton model having frequencies to, for example, 30 Hz, it is possible to obtain a merged (gimbal-free) model with a highest frequency of perhaps 1000 Hz. The following brief discussion attempts to make this physically plausible. At the component level the Craig-Bampton model contains interface matrices and deformation shapes that fully capture the interface flexibility. When the Craig-Bampton model is used for system level analysis its hinge rotation is made free, and one extra mode, the rigid rotation, is added to the modal set. There are still three flexible modes, however. This is shown in Figure 4.



- Total flexibility is present - Flexibility captured by high frequency mode

Figure 4. Cantilever and Pinned Modes of Beam with End Inertia

The rigid and first two flexible modes are inertially dominated by the lumped masses and are orthogonal with respect to these masses with only a very small influence from the inertia at the left end. The highest mode has no "spatial room" to be orthogonal to the first three modes on the basis of the lumped masses, because the "spatial room" has been fully used by the first three modes (three modes : three masses). Therefore, the highest mode must have small displacements of the lumped masses, using them as merely a fine adjustment, and consist mainly of the rotation of the inertia at the left end. This is illustrated by the figure, from which it is clear why the highest modal frequency is so large: its modal mass is derived almost entirely from the local inertia alone, and is very small.

Effect of HGHF Response on Transfer Functions

If the local flexibility at an actuator/sensor interface is large relative to the rest of the structure, and the mass/inertia placed at that interface is very small, the collocated transfer function calculated at the interface will have not only high frequency, but also high gain at high frequency. A large portion of the total flexibility seen directly by the actuator is represented by a high frequency mode of vibration. Change in the mass/inertia affects the frequency of the high gain response but has little effect on the gain itself. A useful plot to illustrate this is one that shows the running sum of modal gains versus frequency, and is shown in Figure 5.

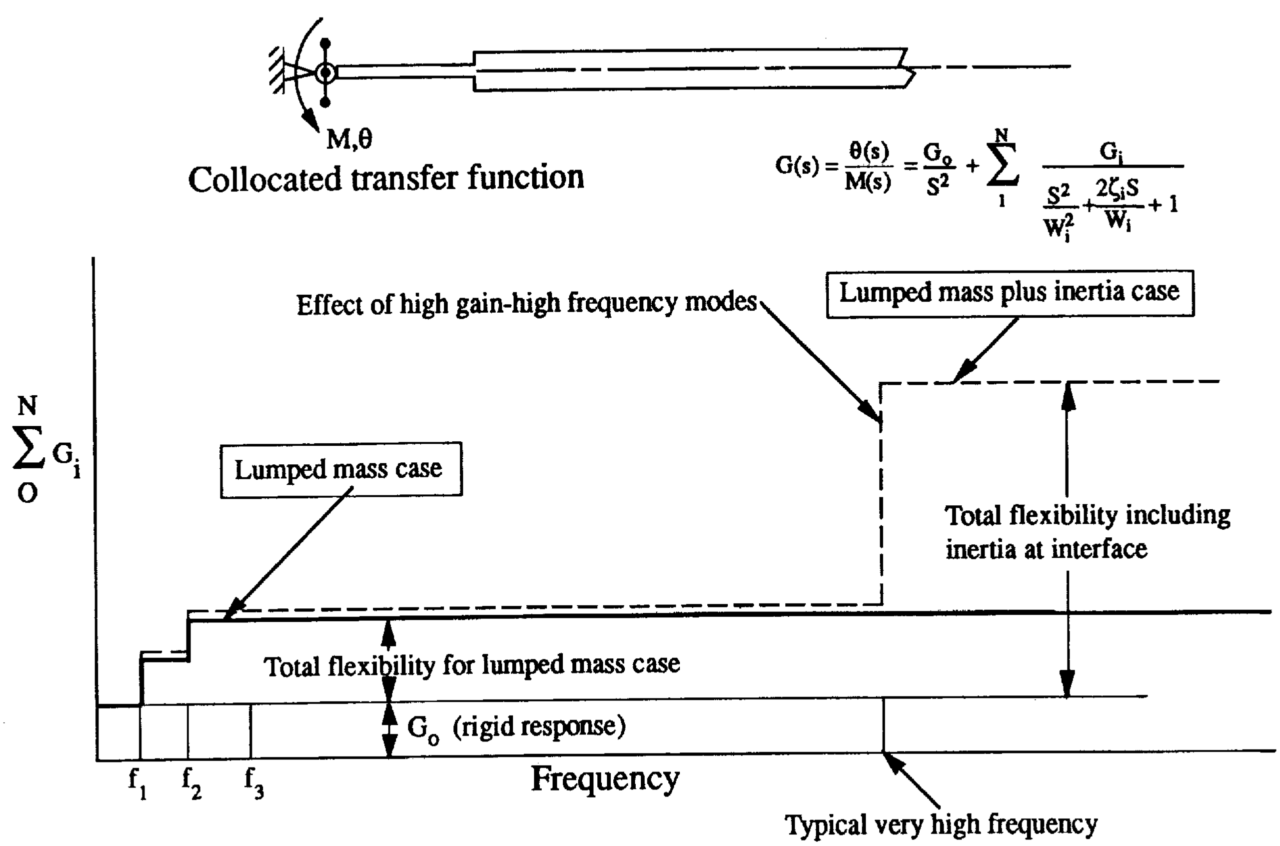


Figure 5. Gain Summation Plot for Pinned-Free Beam

This figure illustrates the effects on transfer functions of local inertia in the structural model. Such a plot typically shows very large gain at high frequency. Omitting the local inertia (with Guyan reduction), or equivalently, truncating the high frequency modes, or equivalently, performing the modal analysis by Generalized Dynamic Reduction, eliminates this gain, resulting in inaccurate control response predictions at all frequencies. This type of plot is a simple check that should always be made for structural model transfer functions to evaluate the presence and importance of HGHF responses associated with a particular hardware application and transfer function. If HGHF modes do occur, their accuracy and effect on control response must be studied. If they do not, either the structural design is very efficient and well adapted to the controls application, or the structural model is deficient for control-structure-interaction simulations (ie. local flexibility may be missing). If HGHF modes are present their effects are important in low frequency as well as high frequency dynamics predictions. This has different consequences for component mode and system mode formulations, as will be discussed later.

The Bode plot in Figure 6 illustrates typical low frequency effects of the presence of HGHF responses for a single-input-single-output (SISO) collocated transfer function.

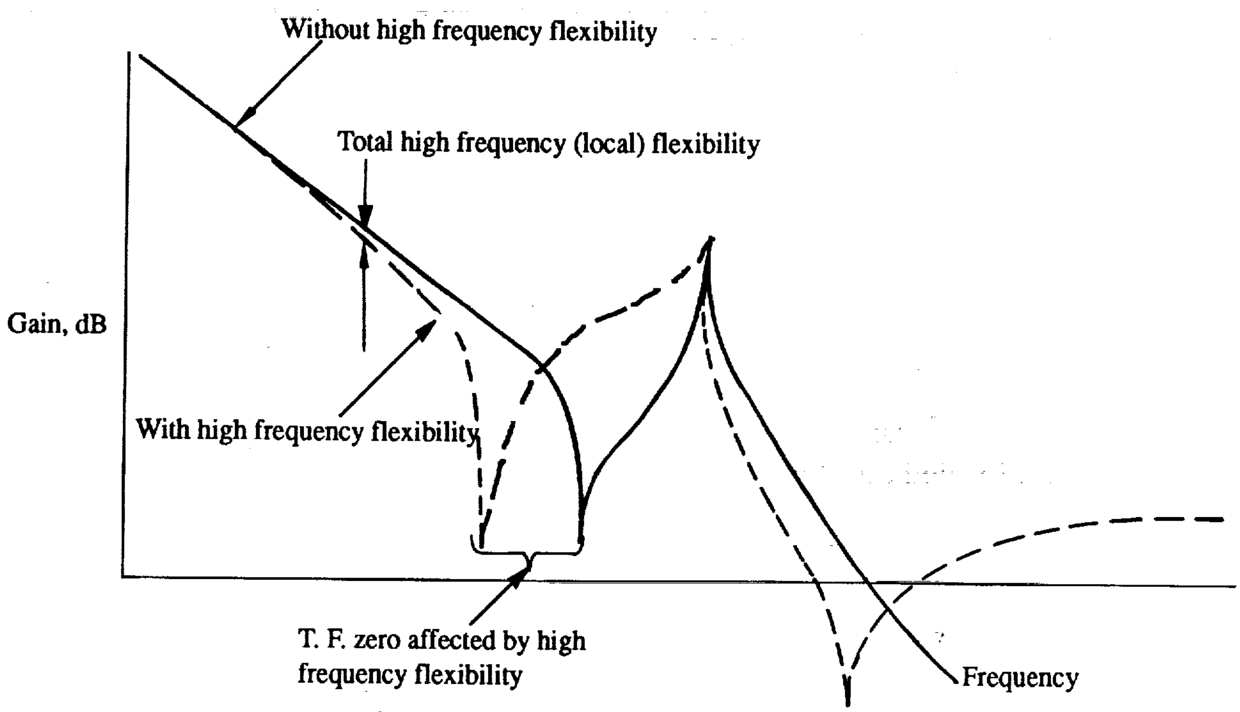


Figure 6. Effect of High Frequency Flexibility on Bode Gain

The effects derive from the placement of the low frequency transfer function zero. Bode plots including high frequency flexibility show that all of the transfer function zeros, particularly the lowest ones, are moved to lower frequencies. In addition, a transfer function zero is produced above the highest retained mode. The low frequency gain is reduced, resulting in loss of agility. The gain is increased between the lowest zero and pole, a region where stability may be in question because of compensator rolloff and phase. Above the first pole, gain is slightly reduced, improving flexible mode stability. Finally, the high frequency flexibility eliminates the structural rolloff customarily seen in truncated modal models. This figure and brief discussion suggest that reliable control-structure-interaction studies require careful attention to local modeling and some means of retaining HGHF responses. Various solution procedure options that effectively eliminate HGHF response should be avoided. These include modal truncation, Guyan reduction, Generalized Dynamic Reduction, and integration schemes that filter out high frequency responses.

Extension of the consequences discussed here for a very simple case to the complicated multi-input-multi-output (MIMO) problem appears to require numerical evaluation for each special case. It appears, however, that important control response consequences are as likely to occur in MIMO as in SISO systems.

Numerical Examples

Examples are given below to help fix the ideas described above. These examples have counterparts that have occurred in practice.

Example 1: The following example is a simplification of a generic cantilevered structure connected to an appendage by a servo mechanism. The actuator/sensor is assumed to be collocated and is represented by a soft spring and inertia. The cantilevered structured is modeled by a beam that has a fundamental frequency of 5.0 Hz when fixed at one end. The appendage is modeled by a beam with a fixed end fundamental frequency of 1.0 Hz. Both beams are finite element models. For simplicity, actuator/sensor hardware is modeled by a single spring and inertia on the cantilevered structure side only. Figure 7 illustrates the model.

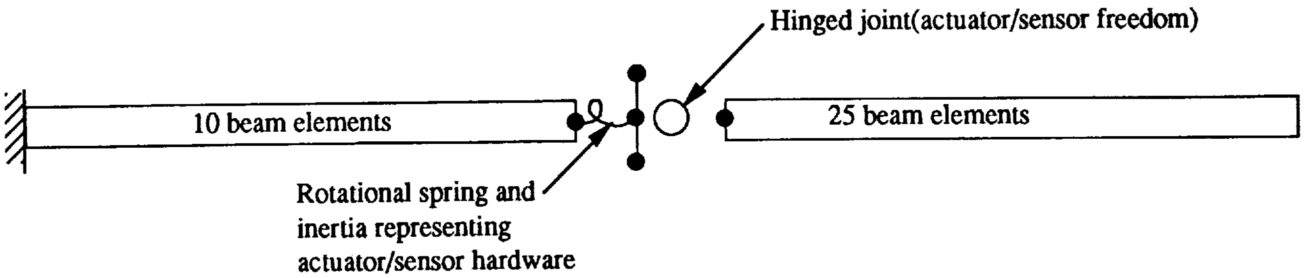


Figure 7. Flexible Beams Connected By Controller

The spring and inertia produce a HGHF mode. The soft interface spring has been chosen such that the gain of the HGHF mode is five times the sum of the gains of the other modes (a factor of five is not uncommon, in practice, for models that have been validated by traditional methods). The local inertia has been chosen to be arbitrarily small.

Figure 8 is a plot of the running sum of the modal gains for the collocated transfer function at the actuator/sensor interface, with and without the HGHF mode.

Figure 9 is a Bode plot of the transfer function with and without the HGHF mode.

The HGHF response tends to swamp the effects of the other modes, causing the Bode gain to remain high as the frequency increases. The low frequency transfer function zeros occur at lower frequencies due to the HGHF response. An important feature is the gain increase just above the lowest zero. This can cause low frequency stability problems.

Example 2: This is a qualitative discussion that refers to the previous example. The text thus far has referred to collocated transfer functions only. The HGHF response problem also applies to non-collocated transfer functions. If the sensor had been chosen to be at the free end of the appendage in the previous example, then care would be required to model local flexibility and inertia at that location. The transfer function would now be affected by at least two HGHF modes. In addition, the previous example assumed actuator flexibility to occur on only the cantilevered structure side. In actuality, there is local flexibility, due to the actuator, on the appendage side also. If all three sources of flexibility were modeled by single springs, there may be up to three HGHF modes each representing a simplification of a portion of the local hardware flexibility.

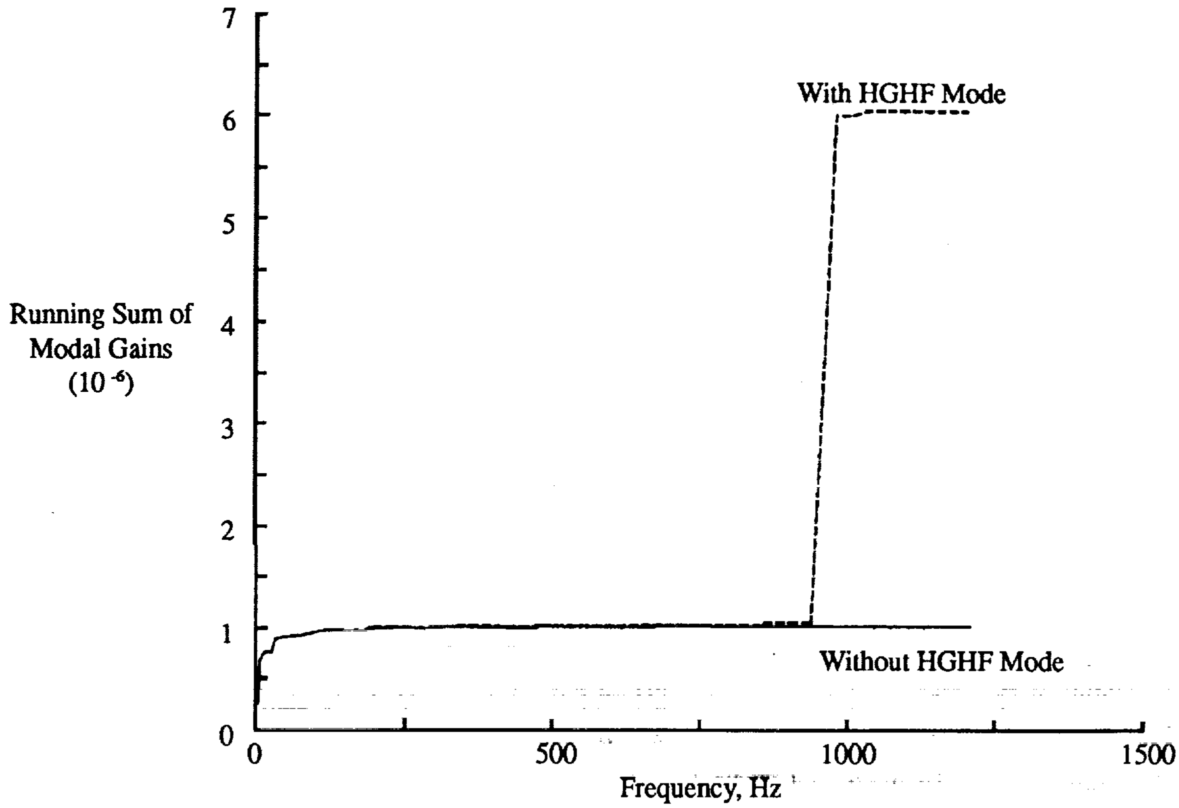


Figure 8. Gain Summation Plot for Cantilever Beam Hinged to Free Beam

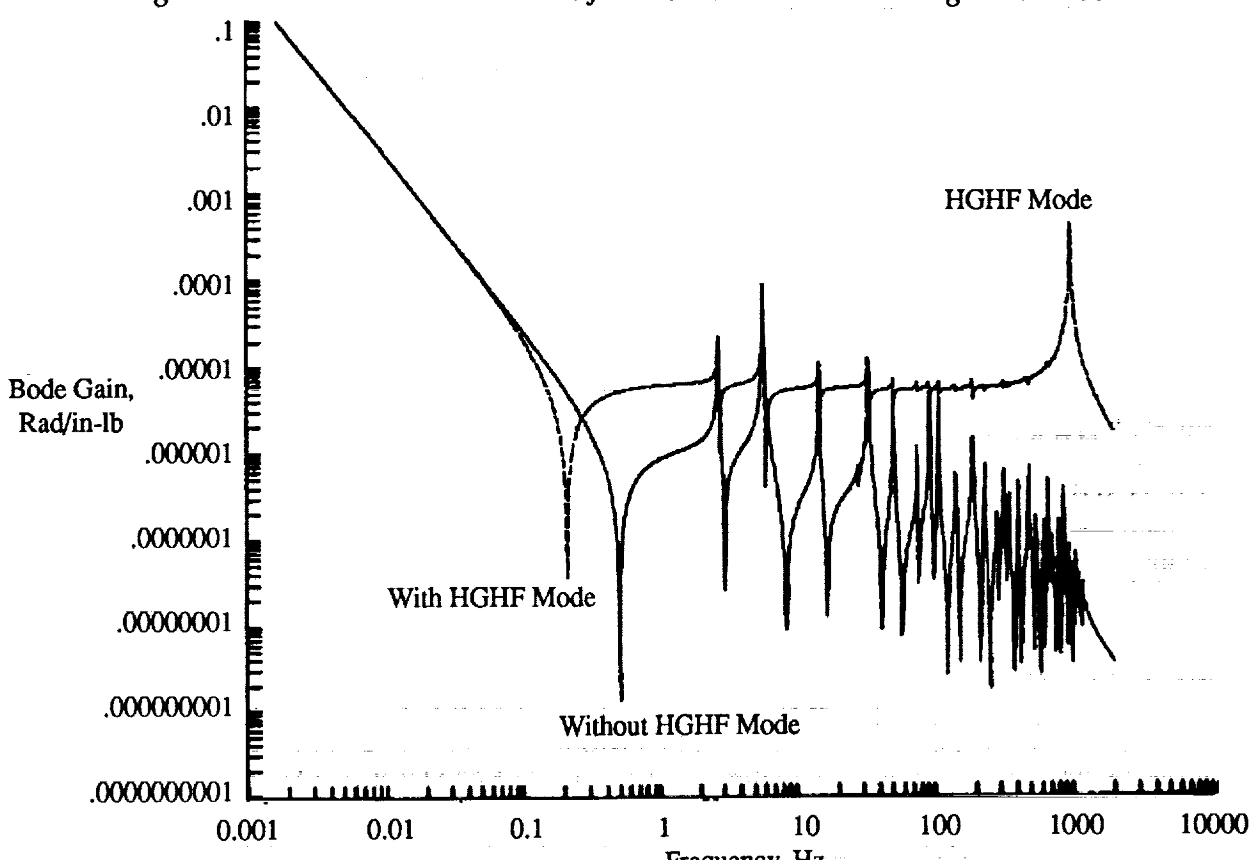


Figure 9. Transfer Function for Cantilever Beam Hinged to Free Beam

Example 3: The previous two examples discussed local flexibilities that were modeled by single rotational springs and inertias. Single spring approximations may be the result of condensing a much more complicated interface flexibility model to a single degree of freedom spring. Figure 10 shows a more refined interface model that produces similar response to that discussed above, but adds important features to the problem.

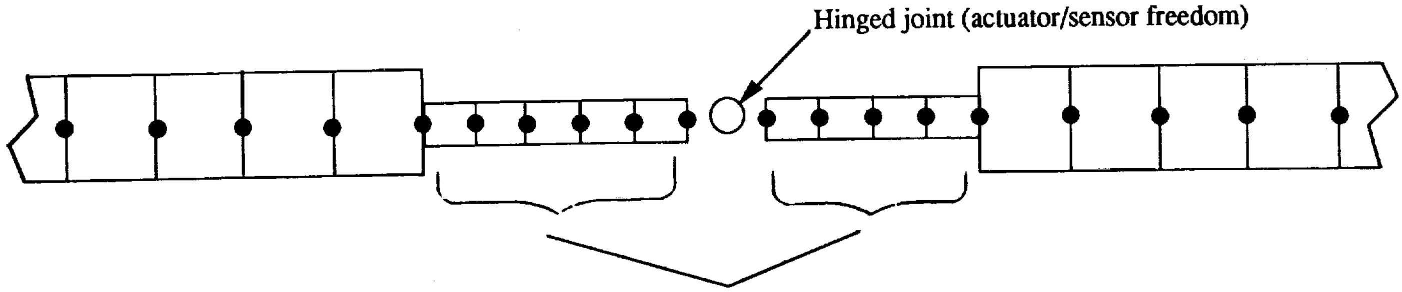


Figure 10. Hinged Beam With Locally Refined Modeling

The modeling includes a series of short (relative to the appendages that are connected by the actuator/sensor interface hardware) flexible beams. In the case of a single rotational spring an inertia had to be added to dynamically capture the local flexibility. Here, provided the short beam elements closest to the interface are not much more flexible than the other beams modeling the interface hardware, modes associated with the interface beams and lumped masses may capture a large portion of the local interface flexibility. Adding inertias will enable the rest. The Figure 11 running sum gain plot shows the effect of interface flexibility and mass/inertia modeling on the transfer function gains.

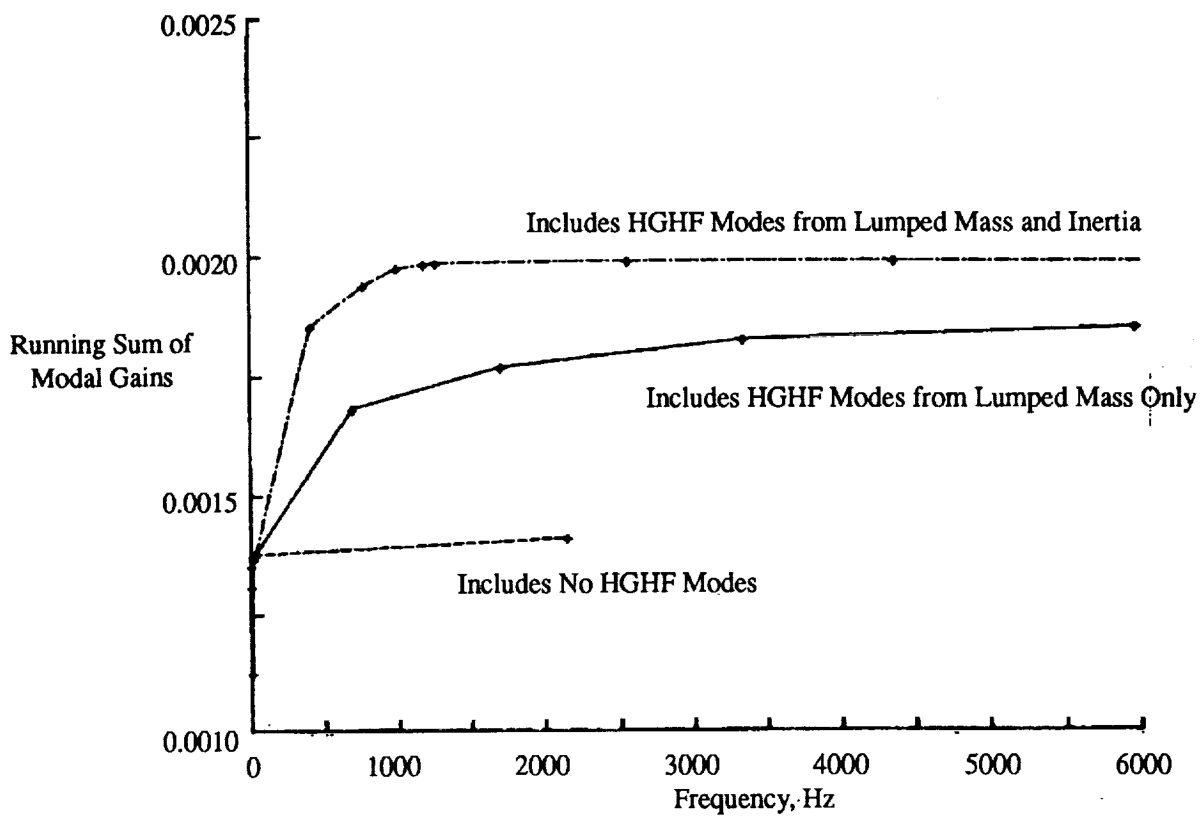


Figure 11. Gain Summation Plot for Beams Hinged at Flexible Interface

Each of the HGHF modes shown on the plot (by the marked points) are associated with local bending modes of the beam model representing the actuator/sensor interface hardware.

Testing to Validate Transfer Function Poles and Zeros

Testing to validate control-structure-interaction models is difficult because for the controls application it is really the system transfer functions, not only the low frequency modal data, that are needed. Unfortunately, tests to validate transfer function poles and zeros are subject to several severe limitations. First, although the best test is a system level transfer function test, the hardware to perform this test is generally not available until late in a program. Thus the data may confirm a model but it is not timely for design purposes. Second, a fixed interface modal survey test may partially confirm a model but it is unlikely that the available fixed interface for testing is truly the actuator/sensor interface. In addition, the torquer may not be available for early testing. If it is available it most likely cannot be physically separated into fixed interface parts as may be desirable in the math model formulation. Even if it could be separated, high frequency data are not practically obtainable in modal survey tests. Despite these real concerns over testing practicality and hardware availability, it is worthwhile to outline an overall testing plan that, if implemented, would address the model validation issue for controls. The testing would begin with fixed interface modal surveys of structural components to determine the overall flexibility of the component tested, up to and including an available fixed interface. This is illustrated by Figure 12.

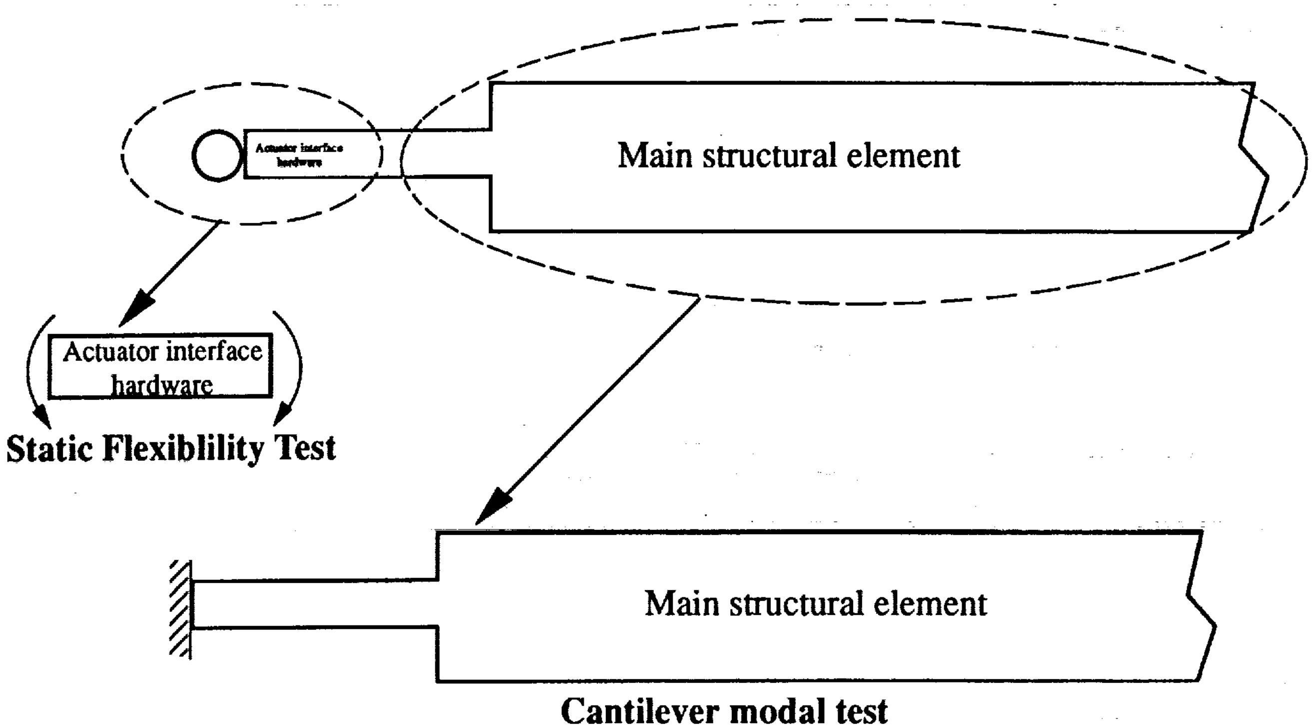


Figure 12. Structural Component Static and Modal Testing Scheme

In order to test the actuator interface hardware it is necessary to supplement component modal testing with static flexibility tests to characterize local flexibility. This will provide data to accurately define flexibility of interface hardware. From these data Craig-Bampton models can be validated and local flexibilities characterized. Component mode dynamic analysis or system modal analysis can proceed from this basis with a reasonable level of confidence. The models should show HGHF responses if careful modeling of both local flexibility and inertia is done to match test data and mass distribution in the interface region.

Finally, system level modal testing is also necessary. This provides low frequency validation of the merging of all of the component models. It approximately describes the low frequency mode shapes and the placement of the low frequency poles. The model resulting from component modal, local static flexibility, and system modal testing may be able to predict system transfer functions in the low frequency range. However, this prediction is a sensitive one. In particular, the system transfer function zeros have not been validated directly and are unlikely to be accurately located. Therefore system level transfer function testing, primarily to validate and adjust zero placement, is a desirable final step in validating structural models for the controls application. Since such testing is difficult and costly, and, as discussed above, may be impractical in some cases, analysis should be done to assess the sensitivity of control response to uncertainty in the low frequency transfer function gain. This analysis is a combined structures and controls endeavor. For sensitive cases, transfer function testing appears an essential final testing step.

HGHF Response Effect on Time Domain Simulations

Provided flexibility and mass have been modeled correctly, system level simulations will almost always include HGHF responses. For component mode simulations this will require integration with very small time steps. This is undesirable, and may even be completely impractical for many problems. A way to avoid this problem by model changes is to add mass and inertia to the hinge interfaces. Whether or not this can produce accurate control response predictions is problem specific. In any case, this appears an undesirable approach. Other approaches necessitate modification of the dynamic analysis methodology.

System mode simulations can escape this problem. These simulations can group the high frequency responses and treat them statically. The total flexibility will be obtained, and the integration can use large time steps. This approach has been very successful in practice.

Component mode methodologies, as presently formulated, do not have this capability. To follow such an approach in component mode analysis it would be necessary to use free-free component modes rather than cantilever or Craig-Bampton modes, and to include residual flexibility of all components as static responses.

Summary

High gain high frequency (HGHF) responses, in dynamic simulations, are the result of small local interface masses and inertias connected to relatively stiff primary structure by flexible elements representing servo mechanisms and their structural attachment schemes. Locally applied forces and torques at control system actuators result in the static response of HGHF modes in addition to dynamic response of low frequency modes.

HGHF responses affect transfer functions by moving all zeros to lower frequencies, particularly those occurring in the low frequency spectrum, and increasing system gain. Excluding local flexibility and local mass/inertia, or equivalently, reducing the modal set via modal truncation or Generalized Dynamic Reduction, suppresses HGHF responses, and can cause inaccurate control-structure-interaction predictions.

Structures and controls engineers need to maintain a close working relationship. An understanding of each others technical tools is necessary to assure accurate modeling of the control-structure-interaction problem. Modeling responsibilities need to be well defined to avoid exclusion or duplication of flexibility, and mass/inertia.

Component modal testing should be supplemented by static flexibility testing of local actuator/sensor hardware since it is unlikely that the available fixed interface, for the component modal test, is actually the actuator/sensor interface. System modal testing should be performed to validate the low frequency poles. If analysis shows the control response to be sensitive to uncertainty in the low frequency transfer function gain, it is desirable to supplement system modal testing with transfer function testing.

If local flexibility and mass/inertia have been modeled correctly, system level simulations will almost always include HGHF responses. System mode simulations can group the HGHF modes statically, thereby retaining the total flexibility and allowing large time steps in time domain analysis. Component mode simulations do not currently have this capability and face difficulties in application to analysis of structures with detailed modeling of actuator/sensor interfaces.

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